AD-A095 910 GENERAL ELECTRIC CORPORATE RESEARCH AND DEVELOPMENT --ETC F/G 20/13
ENGINEERING DESIGN OF HEAT PIPES.(U)
DEC 71 J C CORMAN
UNCLASSIFIED 71-C-271 NL

1 OF 1
AD A
095 910

END
DATE
FILMED
4-81
DTIC

AD A095910

GENERAL ELECTRIC

LEVEL II

GENERAL ELECTRIC COMPANY
CORPORATE RESEARCH AND DEVELOPMENT
Schenectady, N.Y.

DTIC ELECTE MAR 0 3 1981 S D E

ENGINEERING DESIGN OF HEAT PIPES

by

J.C. Corman, Mechanical Engineering Laboratory

Report No. 71-C-271 December 1971

TECHNICAL INFORMATION SERIES

CLASS 1

GENERAL ELECTRIC

General Electric Company
Corporate Research and Development
Schenectady, New York

TECHNICAL INFORMATION SERIES

AUTHOR	SUBJECT	NO
Corman, JC	heat pipe design	71-C-271
		DATE December 1971
TITLE Engineering Design of Heat Pipes,		GE CLASS 1
		NO. PAGES 31
ORIGINATING COMPONENT Mechanical Engineering Laboratory		CORPORATE RESEARCH AND DEVELOPMENT SCHENECTADY, N.Y.

SUMMARY

The design problems resulting from thermal limitations in many industrial products can be alleviated by the proper application of heat pipes. The purpose of this report is to provide engineering design equations and practical guide lines in order to assist product development engineers in evaluating the usefulness of this device. The relationships which are formulated are for the mid-temperature range of heat pipe applications and only a brief discussion is given concerning specific problems which occur when dealing with the extreme temperature ranges. The assembly techniques for heat pipe construction are, to a great degree, tailored to the specific design and only general construction information is supplied.

KEY WORDS

heat transfer, heat pipes, vaporization

INFORMATION PREPARED FOR Mechanical Engineering Laboratory

Additional Hard Copies Available From Corporate Research & Development Distribution P.O. Box 43 Bldg. 5, Schenectady, N.Y., 12301

Microfiche Copies Available From Technical Information Exchange P.O. Box 43 Bldg. 5, Schenectady, N.Y., 12301

RD-54 (10/70)

TABLE OF CONTENTS

NOMENCLATURE

A	Area
C	Constant
c	Specific heat
D	Diameter
F	Force
f	Function
g	Acceleration of gravity
g_o	Gravitational constant
h_{fg}	Latent heat of vaporization
h_{WR}	Wick rise height
H	Enthalpy
h	Heat transfer coefficient
k	Thermal conductivity
K	Wick friction factor
L	Length
M	Momentum
N_{Re}	Reynolds number
P	Pressure
Q	Heat flow
q	Heat flux
R	Inner radius
r	Characteristic radius
T	Temperature
t	thickness
u	Velocity in "x" direction
v	Velocity in "y" direction
W	Flow rate
x	Direction

Greek Letters

Δ	Difference
δ	Wick thickness
σ	Surface tension
Θ	Angle of inclination
ε	Porosity
ψ	Wetting angle
ϕ	r/R
τ_o	Wall shear stress
ρ	Density
μ	Viscosity
ν	Dynamic viscosity

Subscripts

A	Ambient
a	Adiabatic
c	Condenser
ca	Capillary
cf	Coolant fluid
co	Construction
crit	Critical
cw	Container wall
e	Evaporator
eq	Equivalent
ev	Vaporization
f	Friction
fℓ	Fluid
g	Gravity

HP	Heat pipe
i	Inner
ℓ	Liquid
max	Maximum
min	Minimum
o	Outer
p	Pores
r	Radial
sat	Saturation
T	Total
w	Wick
ws	Wick solid
v	Vapor
x	Denotes direction
1	End of evaporator
2	End of condenser

I. INTRODUCTION

Heat pipes have been a focal point of research in recent years and have been exploited in thermal designs for zero gravity applications. However, the thermal transport capabilities of these devices can also be employed to technical and economic advantage in conventional cooling systems.

The thermal advantage derived from heat pipes is achieved by applying two of the most effective heat transport techniques, vaporization and condensation. Although the heat pipe, as a device, is a recent innovation, the transport phenomena which form the functional components have long been studied in the general areas of heat transfer and fluid flow.

The functional process for heat pipes can easily be qualitatively explained. However, the individual processes which are involved; vaporization, condensation, flow in porous structures, and surface tension phenomena, do not lend themselves to closed form mathematical solutions. The coupling of these transport phenomena adds another degree of complexity to the overall solution. The intent of this report is to summarize engineering design relationships which can be employed to evaluate the feasibility of employing heat pipes for specific applications. In most cases, more detailed investigations and/or experimental heat pipe evaluations would have to be performed before a product design would be finalized.

The design relationships described herein are based on "state-of-the-art" information and are primarily concerned with heat pipe performance in the most common, mid-temperature range (-40°F to 300°F). Two items of investigation will be considered; operating temperature distributions and heat input limitations.

Manuscript Received 12/16/71

II. GENERAL CONFIGURATION

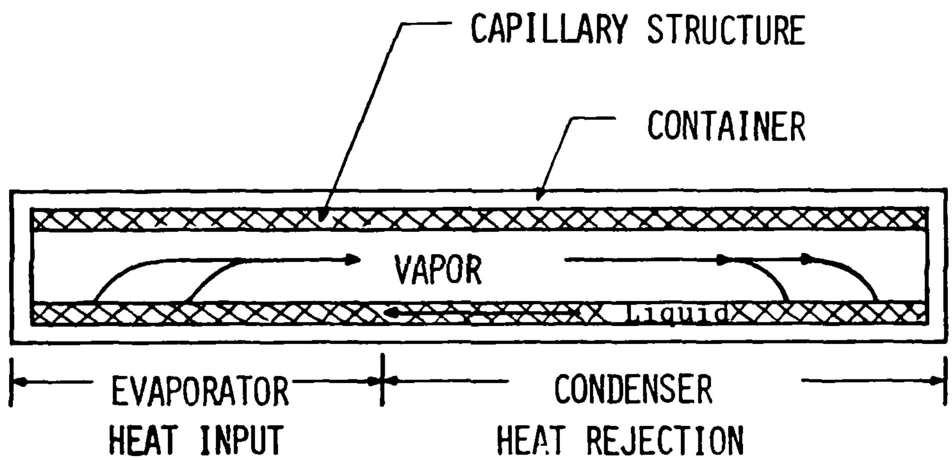
The general configuration is shown in Figure 1. The basic construction consists of a closed container of arbitrary shape, i.e., right circular cylinder, rectangular box, etc. A porous capillary wicking structure is attached to the internal surface of the container. To charge the device, a vacuum is drawn on the container and enough working fluid is introduced into the container to saturate the wick structure and the device is sealed. The container now contains liquid in the wick structure in thermodynamic equilibrium with the vapor in the center portion or vapor space. An equilibrium condition will be reached to correspond with the surrounding temperature. The pressure inside the device will be equal to the saturation pressure for the working fluid at this temperature. Heat pipe working fluids can range from cryogenic liquids to liquid metals, depending upon the temperature range of desired operation. However, the most common applications occur in the mid-temperature range and common fluids such as water, fluoro carbons, alcohol, etc. can be employed.

The heat pipe operates on a closed vaporization-condensation cycle. As heat is added to the heat input section, the liquid in the wick structure is vaporized. The vapor which is formed flows down the vapor-space to the heat rejection section. The vapor condenses on the wick structure as the heat is rejected to a coolant which is external to the container. The liquid condensate now flows back through the porous wick structure to the evaporator section where the cycle is completed. The driving pressure for the fluid flow processes, both vapor and liquid, is supplied by the surface tension phenomena occurring within the capillary pores of the porous wick structure.

In its operating mode, the major temperature drops associated with the thermal transport are due only to: 1) the radial flow of heat into the device, i.e., conduction across the container wall and the evaporation process and 2) the radial flow of heat out of the device, i.e., the condensing process and conduction across the container wall. The temperature drop is therefore dependent upon the heat flux and is independent of length of transport.

FIGURE 1

GENERAL HEAT PIPE CONFIGURATION



The heat pipe will continue to operate in the manner described above until a failure of one of the transport phenomena causes an interruption of the cycle. Failure is most apt to occur because of one of the following:

1) Insufficient liquid supply; an inability of the capillary "pump" to supply sufficient liquid to the evaporator section. The required liquid flow is dependent upon total heat transport in the device (BTU/hr).

2) Ineffective vaporization; a vapor blanketing of the heat input section resulting in a failure of the vaporization process. This is dependent upon input heat flux.

Each of these failures would result in a dry-out of the evaporator section and consequently an excessive temperature rise in the device which is being cooled.

The heat pipe must be designed to avoid the failure modes and to operate within the temperature drop and heat transport requirements as dictated by the specific application. This can be accomplished by altering the type and configuration of the wick structure and the geometrical dimensions of the container or by changing the working fluid.

III. OPERATING LIMITATIONS

The design of the wick structure dictates to a great extent the limiting characteristics of the heat pipe. As previously explained, the two failure modes, wick pumping and wick vaporization, are defined for heat pipes operating in the mid-temperature range.

A. Wick Pumping

The capillary action of the wick structure supplies the driving force for the fluid flow process. This forcing relationship is given by:

$$\Delta P_{ca} = \Delta P_{\ell} + \Delta P_{v} \pm \Delta P_{g} \quad . \tag{1}$$

The pressure drop for liquid and vapor flow is related to the fluid mass flow rate and subsequently to the total heat transport by:

$$W = Q/h_{fg} \quad . \tag{2}$$

Depending on the orientation of the device, gravity either assists or resists the fluid flow process. Since gravity effects on the vapor phase are usually negligible, this component is "lumped" into the liquid phase pressure drop as a body force.

1. Capillary Pressure Rise

Surface tension effects occurring at the liquid-vapor interfaces along the wick length maintain pressure equilibrium between the liquid and vapor. Since liquid is being added to the wick structure at the condenser section in the form of condensate, it is assumed that at this end of the heat pipe the wick is flooded producing an infinite radius of curvature of the interface. As a result, the liquid and vapor pressure are equal. At other locations along the wick structure the difference between liquid and vapor pressure is

$$\Delta P_{ca} = \frac{2\sigma}{r_{ca_{e\alpha}}} \quad , \tag{3}$$

where $r_{ca_{eq}}$ is the equivalent radius of the interface and is given by

$$r_{ca_{eq}} = r_{ca}/\cos\psi \tag{4}$$

At the end of the evaporator section, the liquid-vapor pressure difference is greatest. If more heat is added to the heat pipe, the increase in mass flow rate (Eq. 2) requires greater fluid pressure drops and subsequently higher values of ΔP_{ca}. This process continues until the wick structure no longer sustains the pressure difference and wick dry-out prevents further evaporation at these locations. This limit is defined by a property of the wick structure, denoted by $r_{ca_{min}}$, and must be determined experimentally. Several investigators [1], [2], [3], have determined this quantity by a variety of experimental techniques. The most common of these is the wick rise height, i.e., the vertical distance that liquid rises in a wick structure when working against the force of gravity:

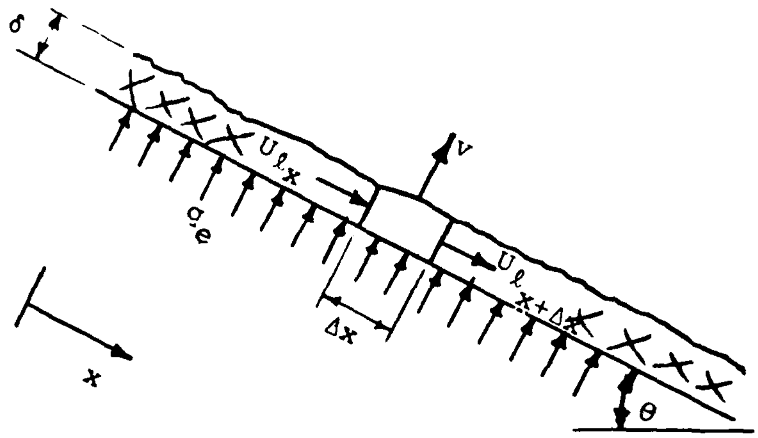
$$r_{ca_{eq_{min}}} = \frac{2\sigma}{h_{WR}\, \rho_\ell \, \frac{g}{g_o}} \ . \tag{5}$$

If the wetting angle, ψ, is not zero in both the wick rise test and the heat pipe, then the experimental value of $r_{ca_{min}}$ must be determined with the same fluid-wick combination as is used in the heat pipe. This insures a proper value for $r_{ca_{eq_{min}}}$.

2. Liquid Flow Pressure Drop

The liquid flow through the porous wick structure creates the largest pressure drop in the system. To analyze this flow process, the wick structure is assumed to be saturated with liquid and the flow is single phase (this assumption will be discussed in more detail in a later section). The liquid velocity is small ($N_{Re} << 1$) and the flow is assumed to obey Darcy's law [4], [5].

The general configuration is shown in Figure 2.



LIQUID FLOW IN WICK STRUCTURES

Figure 2

For the increment Δx, the continuity equation is given by

$$W_{\ell_x} - W_v = W_{\ell_{x+\Delta x}} \tag{6}$$

The velocities are therefore

$$u_{\ell_{x+\Delta x}} - u_{\ell_x} = \frac{-A_v \ v_v \ \rho_v}{\varepsilon \ A_T \ \rho_\ell} \quad . \tag{7}$$

For planar heat pipes and for circular heat pipes in which the wick structure is thin compared to the pipe I.D.

$$A_T \cong \pi \ D_i \ \delta \tag{8}$$

and in the limit as $\Delta x \to 0$

$$\frac{du_\ell}{dx} = - \frac{\rho_v}{\rho_\ell \varepsilon \delta} u_v \ . \tag{9}$$

The forces acting on the increment in the "x" direction are due to pressure difference, gravity and friction. Darcy's law is given by

$$F_f = \frac{k}{g_o} \mu_\ell \ \varepsilon^2 \ u_\ell \ A_T \ \Delta x \quad . \tag{10}$$

The momentum equation

$$\Sigma F_x = M_{x+\Delta x} - M_x \ , \tag{11}$$

derived for the increment Δx is

$$\varepsilon A_T (P_x - P_{x+\Delta x}) + \rho_\ell A_T \varepsilon \Delta x \frac{g}{g_o} \sin\Theta - \frac{K\mu_\ell}{g_o} \varepsilon^2 A_T u_\ell \Delta x$$

$$= \frac{\rho_\ell}{g_o} \varepsilon A_T \ (u^2_{\ell_{x+\Delta x}} - u^2_{\ell_x}) \ , \tag{12}$$

or in the limit

$$- \frac{dP}{dx} + \rho_\ell \frac{g}{g_o} \sin\Theta - \frac{K\mu_\ell}{g_o} u_\ell \varepsilon = \frac{\rho_\ell}{g_o} \frac{d}{dx} (u_\ell^2) \tag{13}$$

The energy equation is similarly derived for this increment and if conduction is neglected

$$H_\ell \frac{du_\ell}{dx} + \frac{u_v \; \rho_v \; H_v}{\rho_\ell \; \varepsilon \; \delta} = \frac{q}{\rho_\ell \; \varepsilon \; \delta} \tag{14}$$

The energy and continuity equations are combined to give

$$\frac{du_\ell}{dx} = - \frac{q_e}{h_{fg} \; \rho_\ell \; \varepsilon \; \delta} \tag{15}$$

This equation implies that the heat input is constant along the evaporator. Since the liquid velocity is zero at the end of the evaporator, the variation of liquid velocity with axial dimension is linear

$$u_\ell = - \frac{q_e}{h_{fg} \; \rho_\ell \; \varepsilon \; \delta} \, x \; . \tag{16}$$

The momentum equation is now solved for the evaporator and the pressure difference is

$$\Delta P_e = P_e - P_1 = \frac{1}{g_o} \frac{q_e^2 \quad L_e^2}{h_{fg}^2 \rho_\ell \; \varepsilon^2 \; \delta^2} - \frac{K\mu_\ell}{g_o} \quad \frac{q_e}{h_{fg} \; \rho_\ell \; \delta} \frac{L_e^2}{2}$$

$$- L_e \; \rho_\ell \frac{g}{g_o} \sin\Theta \; . \tag{17}$$

A similar solution is obtained for the condenser section and is based on evaporator heat input flux:

$$\Delta P_c = P_2 - P_c = - \frac{1}{g_o} \frac{q_e^2 \; L_e^2}{h_{fg}^2 \; \rho_\ell \; \delta^2 \varepsilon^2} - \frac{K\mu_\ell}{g_o} \frac{L_e \; q_e}{h_{fg} \; \rho_\ell \; \delta} \frac{L_c}{2}$$

$$- \rho_\ell \; L_c \frac{g}{g_o} \sin\Theta \; . \tag{18}$$

If an adiabatic section exists between the evaporator and condenser, this pressure drop is given by

$$\Delta P_a = P_1 - P_2 = -\rho_\ell \frac{g}{g_o} L_a \sin\Theta - \frac{K\mu_\ell}{g_o} \frac{L_e q_e}{h_{fg}\rho_\ell \delta} L_a \ . \tag{19}$$

The total liquid pressure drop is

$$\Delta P_\ell = \Delta P_e + \Delta P_c + \Delta P_a \ , \tag{20}$$

and is given by

$$\Delta P_\ell = P_e - P_c = - \frac{K\mu_\ell}{g} \frac{q_e L_e}{h_{fg} \rho_\ell \delta} \left(\frac{L_e}{2} + L_a + \frac{L_c}{2}\right)$$

$$- \rho_\ell \frac{g}{g_o} \sin\Theta \left(L_e + L_c + L_a\right), \tag{21}$$

The pressure drop in the wick structure is related to another property of the wick, K, which is also obtained by experimental methods [1], [6], [7].

3. Vapor Flow Pressure Drop

The pressure in the vapor region is greatest in the evaporator section and decreases as the vapor flows to the condenser section. In the evaporator section, vapor is continuously being added to the flow stream, thereby producing an injection flow. In the condenser section vapor is being removed from the stream, thereby producing a suction flow.

This type of flow process, injection or suction, lends its self readily to analysis and a significant amount of analytical work has been done in this area [8], [9], [10]. A simplified analysis is sufficient for calculating the vapor flow pressure drop because, in the mid-temperature range, the vapor flow pressure drop is small in comparison to the liquid flow pressure drop. Furthermore, conditions which can be limiting for liquid metal heat pipes such as sonic vapor velocity and liquid entrainment are not as critical for lower temperature working fluids.

For the evaporator section, the injection flow is characterized by Figure 3, where V_{o_e} represents the constant injection velocity of the vapor, and r = R is the wick-vapor interface. For a circular tube the continuity equation is

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial r} + \frac{V}{r} = 0 \tag{22}$$

Using an integral approach, the momentum equation is

$$\frac{dP}{dx} - \frac{2\mu_v}{g_o R} \left.\frac{\partial u}{\partial r}\right|_{r=R} = \frac{\rho_v}{R^2 g_o} \int_o^R \frac{\partial}{\partial x} (u^2)\ rdr \quad . \tag{23}$$

The boundary conditions are

$$\begin{aligned} \text{at } r=R \quad & v = -v_{o_e} \\ r=0 \quad & v = 0 \\ & \frac{du}{dr} = 0. \end{aligned}$$

Now assuming that v is not a function of x,

$$v = f(r) = f, \tag{24}$$

and from continuity

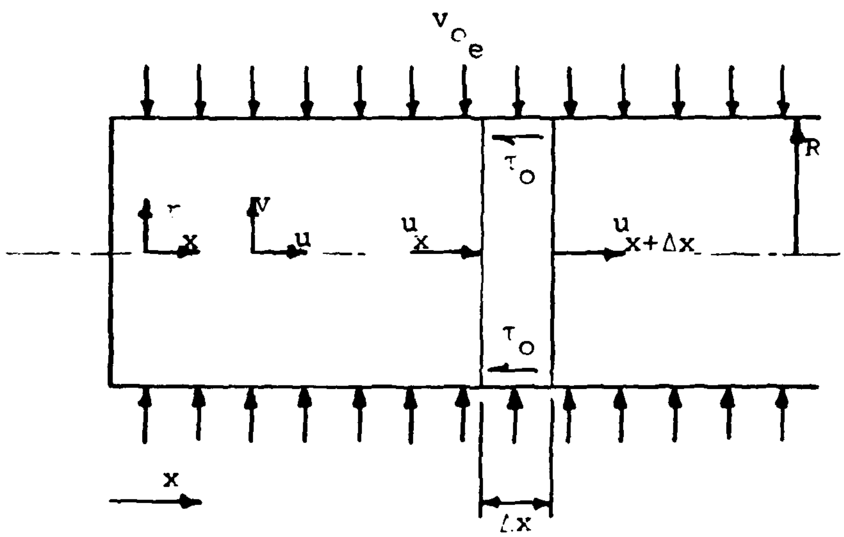
$$u = -f'x - \frac{f}{r} x \ . \tag{25}$$

Assuming a third order polynomial fit for the function f and matching the boundary conditions gives

$$v = v_{o_e} \ (\phi^3 - 2\ \phi), \tag{26}$$

$$u = -4\ v_{o_e} \frac{x}{r} (\phi^2 - 1), \tag{27}$$

where $\phi = \frac{r}{R}$.



VAPOR FLOW IN VAPOR SPACE

Figure 3

Substituting the velocity expression into the momentum equation and integrating from $x = 0$ to $x = L_e$

$$P_1 - P_e = \frac{\rho_v v_{o_e}^2}{2g_o} - \left[\frac{16}{3}\left(\frac{L_e}{R}\right)^2 - 16\,\frac{\nu_v}{v_{o_e} R}\left(\frac{L_e}{R}\right)^2\right] . \qquad (28)$$

Similar expressions can be derived for the condenser section and the adiabatic section. Assuming a constant radius tube and constant vapor density, the boundary velocities are related by

$$v_{o_e} L_e = v_{o_c} L_c .$$

The end-to-end vapor flow pressure drop is given by

$$P_e - P_c = \frac{\rho_v v_{o_e}^2}{2g}\left[\left(\frac{L_e}{R}\right)^2\left(\frac{16\,\nu_v}{v_{o_e} R}\right)\left(1 + \frac{L_c}{L_e} + \frac{L_a}{L_e}\right)\right] . \qquad (29)$$

The injection velocity, v_{o_e}, is related to the heat input by

$$v_{o_e} = \frac{Q_e}{2\pi h_{fg}\,\rho_v\,R\,L_e} . \qquad (30)$$

The above relationship holds only for low injection rates. Cotter [11] gives a relationship for vapor flow pressure drop in the limit where the radial Reynolds number, N_{Re}, approaches infinity (N_{Re_r} is defined as $N_{Re_r} = \frac{Rv_{o_e}}{\nu_v}$)

$$P_e - P_c = (1 - 4/\pi^2)\,\frac{Q_e^2}{8\rho_v R^4 h_{fg}^2 g_o} . \qquad (31)$$

For a conservative estimate of vapor flow pressure drop, the maximum value from Equation (29) or (31) can be used.

4. Wick Pumping Limitation

If the heat input to the heat pipe is restricted by wick pumping, the capillary pressure rise assumes its limiting value. This value, given by Equation (3) along with the liquid flow pressure drop determined from Equation (21) and the vapor flow pressure drop determined from Equation (29) are substituted into Equation (1). The limiting value for evaporator heat flux input (based upon tube O.D.) is

$$q_{e_{max}} = \frac{\dfrac{2\sigma}{r_{ca_{min}}} - \rho_\ell \dfrac{g}{g_o} L_e \sin\Theta\left(1 + \dfrac{L_a}{L_e} + \dfrac{L_c}{L_e}\right)}{\dfrac{K L_e{}^2 \nu_\ell}{2h_{fg} g_o \delta}\left(1 + 16\dfrac{\nu_v}{\nu_\ell}\dfrac{\delta}{KR^3}\right)\left(1 + \dfrac{L_c}{L_e} + 2\dfrac{L_a}{L_e}\right)} \qquad (32)$$

In most heat pipe applications, the vapor flow pressure drop can be neglected. Furthermore, for horizontal applications, the angle Θ is zero. With these two conditions, Equation (32) reduces to

$$q_{e_{max}} = \frac{4\sigma\, h_{fg}\, g_o\, \delta}{r_{ca_{min}}\, K\, L_e^2\, \nu_\ell \left(1 + \dfrac{L_c}{L_e} + \dfrac{L_a}{L_e}\right)} \qquad (33)$$

The heat pipe operation is dependent upon a grouping of

1) Fluid parameters

$$N_{f\ell} = \frac{\sigma\, h_{fg}}{\nu_\ell}\ , \qquad (34)$$

2) Wick parameters

$$N_w = \frac{1}{r_{ca_{min}} K}\ , \qquad (35)$$

3) Construction parameters

$$N_{co} = \frac{\delta}{L_e (L_e + L_c + 2L_a)}\ . \qquad (36)$$

As previously indicated, the wick parameters must be determined experimentally. In order to successfully predict heat pipe performance, the wick parameters must be determined under conditions which simulate heat pipe operation. The non-uniformity of wick structures partially explains the discrepancy in these values when presented in the open literature. This also indicates the need for testing wick samples before they are employed in prototype heat pipes.

A representative listing of wick parameters is presented in Table I. The more common mid-temperature working fluid properties are presented in Table II.

B. Departure from Vaporization

The heat introduced at the evaporator section of the heat pipe is removed by vaporizing the working fluid. If the capillary pumping action is sufficient to supply the liquid, this process continues. However, as in smooth surface boiling, a vapor blanketing, i.e., dry out, condition can occur where the point of occurrence is a function of the local heat flux rather than the bulk liquid supply. On smooth surfaces this phenomena has been extensively investigated [12], [13], however, when an irregular porous structure is placed on the smooth surface, the vaporization process is altered and little information exists to define the qualitative or quantitative results of this process.

In the wick vaporization process, vapor is generated within the wick structure [14], [15], [16], [17], and must be vented through the wick. It has been reported [18] that if proper vapor venting is permitted, the heat flux characteristic of departure from vaporization for a wick covered surface is greater than the dryout heat flux for a smooth surface. The wick structures employed in conventional heat pipe construction are highly porous ($\varepsilon > 70\%$) and relatively thin ($\delta < 40$ mils). For this class of wick structure, a smooth surface, pool boiling, correlation, such as Kutateladze's equation [19]

$$q_{crit} = \sqrt{C}\ [h_{fg} \sqrt{\rho_v}\ \sqrt[4]{\sigma g_o (\rho_\ell - \rho_v) g}\ ], \qquad (37)$$

where $\sqrt{C} = 0.4$, will give a conservative estimate for the departure from vaporization heat flux limit.

TABLE I

RANGE OF WICK PARAMETERS [7]

Wicks Type	Material	Porosity	$r_{ca_{min}}$ ft.	K 1/ft²
Felt	Nickel	.69	1.2×10^{-4}	61.5×10^{8}
Felt	Nickel	.92	3.1×10^{-4}	1.7×10^{8}
Felt	Copper	.90	7.5×10^{-4}	$.75 \times 10^{6}$
Foam	Nickel	.94	7.5×10^{-4}	$.34 \times 10^{8}$
Screen	Nickel	200 mesh	2.1×10^{-4}	12.1×10^{8}
Screen	SS	200 mesh	1.9×10^{-4}	17.9×10^{6}

TABLE II

FLUID PROPERTIES

Fluid	Temperature °F	Pressure psia	ρ_ℓ #/ft³	σ #/ft	h_{fg} BTU/#	ν_ℓ ft²/hr	ν_v ft²/hr
Water	120	1.7	61.2	.0047	1026	.022	4.75
Ammonia	120	286	35.2	.001	455	.0128	.028
Freon 12	120	172	75.9	.00041	53	.0054	.0072
Freon 22	120	277	68.1	.00033	65	.0053	.006
Ethyl Alcohol	120	3.9	48.5	.0014	258	.0035	.047

In most heat pipe applications, the heat flux as defined by the pumping limit, Equation (32), will be much less than the departure from vaporization limit as defined by Equation (37). In this situation, the heat transferred by the heat pipe is limited by the capillary liquid supply. However, in applications in which the heat pipe's evaporator heat flux approaches the limit as defined by Equation (37) and/or if the wick properties differ significantly from those previously stated ($\varepsilon > 70\%$, $\delta < 40$ mils) independent verification of the departure for vaporization limit should be obtained before design is fixed.

C. Dependency of Wick Pumping Upon Wick Vaporization

The wick pumping limit as defined by Equation (32) is based upon single-phase fluid (liquid) flow within the wick structure. The liquid flow is in the axial direction. However, as stated in the preceeding section, vapor is generated within the wick structure in the evaporator section. This vapor flows radially through the wick structure to the vapor space. The liquid flow process is thereby disrupted by this vapor flow because the vapor flow passages occupy liquid flow area. This occupancy can significantly reduce the heat flux limit as predicted by single-phase relationships. Again, however, if the wick structures are relatively thin and porous ($\varepsilon > 70\%$, $\delta < 40$ mils), the vapor is vented through the wick structure without significant reduction in predicted performance [14].

IV. OPERATING TEMPERATURES

A. Temperature Differences

In the vapor space, the only end-to-end temperature variations are associated with changes in vapor pressure due to vapor flow pressure drops. As indicated in a previous section, these pressure drops are very small, and the vapor space is assumed to operate at constant saturation pressure and temperature along the axial length of the heat pipe. However, because heat is transferred radially into and radially out of the heat pipe, temperature drops occur. In steady state operation, the total heat added to the evaporator section must be rejected at the condenser section.

1. Evaporator Section

In the evaporator section, the thermal resistances which account for the temperature drops are the wall conduction resistance and vaporization resistance. Assuming a planar geometry the temperature drop through the wall is

$$\Delta T_{cw} = T_{cw_o} - T_{cw_i} = \frac{q_e}{k_{cw}/t_{cw}} \; . \tag{38}$$

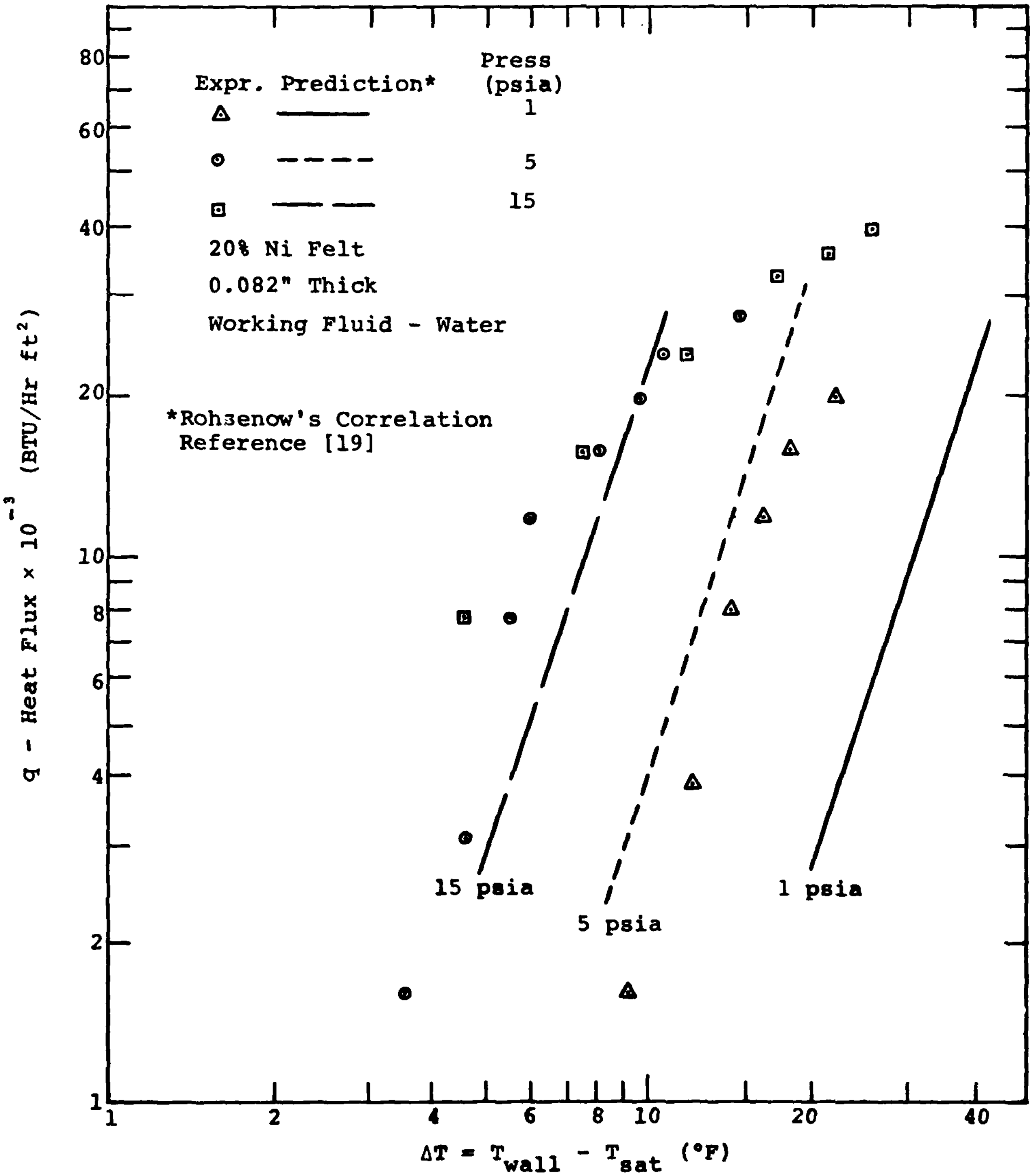
As previously indicated, no explicit equation exists to define the non-linear transport phenomena or resulting heat transfer coefficient for wick vaporization. However, the thermal resistance from the process is not controlling and can be conservatively approximated as shown in Figure 4 [14] by Rohsenow's correlation [19] for smooth surface pool boiling:

$$\Delta T_e = T_{cw_i} - T_{sat} = C \frac{h_{fg}}{c_\ell} \left(\frac{q_e}{\mu_\ell h_{fg}} \sqrt{\frac{g_o \sigma}{g(\rho_\ell - \rho_v)}} \right)^{0.33} \left(\frac{c_\ell \mu_\ell}{k_\ell} \right)^{1.7} \tag{39}$$

where C is dependent upon the surface and ranges from .006 to .014.

2. Condensing Section

The vapor is condensed on the inner wick surface, releasing the latent heat of condensation. The heat must then be conducted through the wick structure and container wall to the outer container wall surface.



WICK VAPORIZATION

Figure 4

The thermal resistance associated with the condensing process is so small that it is neglected. The temperature drop associated with the conduction process through the wick and container wall is

$$\Delta T_c = T_{sat} - T_{cw_o} = \frac{q_c}{k_w/\delta + k_{cw}/t_{cw}}. \qquad (40)$$

(The conductivity of the wick structure will be defined in a following section).

3. Total Heat Pipe Temperature Difference

The heat pipe temperature difference is

$$\Delta T_{HP} = T_{cw_{o_e}} - \Delta T_{cw_{o_c}} = \Delta T_{cw} + \Delta T_e + \Delta T_c \qquad (41)$$

For standard heat pipe construction, the largest temperature difference and the controlling thermal resistance occurs due to conduction across the wick-liquid matrix at the condensing section.

B. Saturation Temperature

If a constant heat input is applied to the evaporator section, and the heat is rejected from the condenser section to a coolant fluid at T_A, the thermal resistances in the condenser section control the saturation temperature of the heat pipe. The temperature difference between the saturation temperature and the ambient temperature is the result of; 1) the thermal resistance of conduction process as defined by Equation (40) and 2) the thermal resistance due to the film heat transfer coefficient of the coolant fluid. The saturation temperature is therefore

$$T_{sat} = T_A + \frac{q_c}{k_w/\delta + k_{cw}/t_{cw} + h_{cf}} \quad . \qquad (42)$$

A standard heat pipe always operates in a thermodynamic equilibrium condition. The heat pipe pressure (saturation pressure) is defined by the saturation temperature and the equilibrium temperature-pressure relationship for the heat pipe working fluid.

C. Wick Conductivity

The thermal studies of porous media [20], [21], [22], [23], have not resulted in a universal analytical expression to define the thermal conductivity of the composite as a function of the thermal conductivities and composition of the constituents. The bounds on the value of thermal conductivity are determined [24] by assuming the constituents (liquid and solid) take on a parallel (lower bound)

$$\frac{k_w}{k_{ws}} = \frac{1}{1 + \varepsilon\ (\frac{k_{ws}}{k_\ell} - 1)} \tag{43}$$

or a series (upper bound)

$$\frac{k_w}{k_{ws}} = 1 - \varepsilon\ (1 - \frac{k_\ell}{k_{ws}}) \tag{44}$$

arrangement in the heat flow direction. An empirical correlation which was derived from a great volume, [25], of experimental data is given as

$$\frac{k_w}{k_\ell} = (\frac{k_{ws}}{k_\ell})^{0.28-0.57(\log_{10}(\varepsilon\ \frac{k_{ws}}{k_\ell}))} \tag{45}$$

The relationship given by this empirical equation yields values close to those of the lower bound (43) and also matches the experimental heat pipe data concerning temperature difference across the condenser section.

The thermal conductivity of wick structures varies widely with the type of structure and the construction techniques of the heat pipe. Since the lower bound value gives conservative results this value is recommended unless experimental data is available to more precisely define the thermal conductivity for the particular combination of wick structure and working fluid.

V. EXTREME TEMPERATURE RANGE

The design criteria presented in this report are intended for use in the mid-temperature, heat pipe, application range, however, much work has been done to design heat pipes which operate at extreme temperatures and which employ liquid metals and cryogenic fluids as working fluids. Heat pipe operation at these extremes encounters additional design constraints the definition of which are beyond the scope of this report.

A. Liquid Metals

Much of the early work on heat pipes was initiated by the possible application of liquid metal heat pipes on thermionic devices [26], [27], [28]. The working fluid for these devices has been sodium, potassium, lithium, silver, etc.

The liquid metal working fluids have the advantage of very high surface tension. The heat pipes have much higher wick "pumping" capabilities and therefore are; 1) much less dependent upon orientation with respect to gravity and 2) capable of having extended length. The high thermal conductivity of the working fluid enables the evaporator to be operated without vaporization occurring within the wick structure and also with lower conduction temperature drops in the condenser section. Thus the heat pipes operate at a more uniform end-to-end temperature differential.

Liquid metal heat pipes also have limitations not encountered in heat pipes operating with more conventional fluids:

1) Start-up; The working fluid is in the frozen state at ambient temperatures, therefore the start-up must be accurately controlled to prevent premature burnout [29], [30], [31].

2) Sonic limits; The metal vapor has a lower density and results in higher vapor velocities with the possibility of restricted heat input due to sonic vapor velocity [32], [33].

3) Entrainment; The high interfacial velocities between the liquid being returned within the wick structure and the vapor flow in the vapor space result in entrainment of the liquid into the vapor stream and a reduction in the amount of liquid that is returned to the evaporator section [34], [35].

B. Cryogenic Fluids

The work on low temperature heat pipes has not been as extensive [36], [37], [38]. Although the experimentation with this class of working fluids is much more difficult and experimental data is not readily available, the design limitations seem to be very similar to those encountered in the mid-temperature heat pipes. The extrapolation of empirical data taken with "room temperature" fluids is very questionable but for lack of more complete information it can function as an order-of-magnitude design base.

VI. TEMPERATURE CONTROL

The unique characteristic of a heat pipe as a heat transfer device is that the heat input section is maintained at a uniform temperature along the evaporator length. Attempts have also been made to maintain a uniform temperature throughout a range of varying heat inputs. To perform this function, changes are made to the external heat rejection mechanism or to the internal heat pipe construction.

A. External

As indicated in a previous section, the saturation temperature of the heat pipe is a direct function of; 1) the heat input, 2) the coolant fluid temperature, and 3) the coolant heat transfer coefficient. With all other parameters remaining equal, as the heat input is increased, the saturation temperature increases. If a control on the coolant fluid temperature and/or coolant fluid flow rate (i.e., the heat transfer coefficient) is introduced, and governed by the heat input, a uniform evaporator section temperature throughout a range of heat inputs can be obtained. This is a non-passive control system.

B. Internal

Several internal techniques have been suggested as passive temperature controls for variable, heat input, heat pipes. It is felt, however, that these techniques have not yet been standardized and introduce too many unknowns into the construction and design of mid-temperature range heat pipes.

1) Non-condensable Gas

The introduction of a non-condensable gas into the heat pipe makes a portion of the heat pipe's condenser section non-operable [39], [40], [41], [42]. The saturation temperature is therefore controlled by varying the condenser area as the heat input is varied. This technique has been successfully employed with liquid metal heat pipes, where a sharp interface exists between the liquid metal vapor and the non-condensable gas. Although much experimentation has been conducted on heat pipes with the mid-temperature range fluids, several problem areas exist:

1) the non-condensable gas diffuses from the reservoir into the total condenser section, thus affecting the condensing heat transfer coefficients; 2) the vapor-gas interface is not sharply defined and it's motion is difficult to predict; 3) the gas can enter the wick structure, thus producing a very rapid rise in the wick friction factor and consequently in the liquid flow pressure drop.

2) Two Liquids

Some experimental and analytical investigations [43], [44], have indicated that by employing two liquids of different boiling points in the heat pipe as working fluids, temperature control can be achieved. Although this technique doesn't have some of the inherent disadvantages of the non-condensable gas control, it has not as yet been developed to a reliable application design stage.

VII. COMPATIBILITY AND CONSTRUCTION

The compatibility problem between the heat pipe working fluid and the container vessel is much more severe in liquid metal heat pipes than in the mid-temperature range devices [45], [46]. In the latter heat pipes [47], [48], [49], the most important consideration is to avoid the generation of non-condensable gases. This would usually occur in a reaction in which hydrogen is generated. The effect of this gas liberation is a deterioration of heat pipe performance with time. With some combinations of materials, e.g., water and aluminum, this can occur within a matter of weeks.

As stated in the general discussion, the construction details of a heat pipe are relatively simple. However, as in many produdcts, expertise in construction comes only through familiarization with the assembly techniques. Several general construction concerns which should be kept in mind are as follows:

1) The container must be vacuum tight and must be well evacuated (to approximately 5-15 millitorrs) before charging with the working fluid;
2) The working fluid must be deaerated;
3) The wick structure must be wettable with the working fluid;
4) The wick structure must be in close contact with the evaporator wall;
5) An attempt should be made to use common materials throughout the heat pipe, e.g., copper container and copper wick structure.

These general construction details will not guarantee success of the device but will eliminate the more obvious causes of heat pipe failures.

VIII. CONCLUSION

The heat pipe has the potential of performing a very important thermal transport function in many industrial applications. The technical work and research efforts in this area have not as yet progressed to a stage where a prototype heat pipe design can be made solely from analytical studies. However, enough information is available to carry out an engineering design and trade-off study to identify the economic and technical improvements gained by employing this type of coolant device. The finalized design will require empirical inputs and also experimental evaluation.

REFERENCES

1. Kunz, H.R., et al., "Vapor-Chamber Fin Studies", NASA Report CR-812, June 1967.
2. Farran, R.A., Starner, K.E., "Determining Wicking Properties of Compressible Materials for Heat Pipe Applications," ASME Aviation and Space, June 1968, pp. 659-670.
3. Farrell, J.K., et al., "A Study of the Operating Characteristics of the Heat Pipe", Parts I and II, AEC Report ORO-3411-12, April 30, 1969.
4. Scheidegger, A.E., "The Physics of Flow Through Porous Media", The MacMillan Company, New York, 1960.
5. Luikov, A.V., "Heat and Mass Transfer in Capillary-Porous Bodies", Pergamon Press, 1966.
6. Freggens, R.A., "Experimental Determination of Wick Properties for Heat Pipe Applications", 4th Intersociety Energy Conversion Engineering Conference, September 1969, Paper No. 699108.
7. Phillips, E.C., "Low Temperature Heat Pipe Research Program", NACA Report CR-66792.
8. Yuan, S.W., Finkelstein, A.B., "Laminar Pipe Flow with Injection and Suction Through a Porous Wall", Trans. ASME, 78, p. 719, May 1956.
9. Horton, T.E., Yuan, S.W., "Laminar Flow in the Entrance Region of a Porous-Wall Channel", Applied Science Research, Section A, Vol. 14, 1964-65.
10. Knight, B.W., McIntever, "Laminar Incompressible Flow in Channels with Porous Walls", Los Alamos Scientific Laboratory Report LADC-5309.
11. Cotter, T.P., "Theory of Heat Pipes", Los Alamos Scientific Laboratory Report LA-3246-MS, March 1965.
12. Kutateladze, S.S., "Heat Transfer in Condensing and Boiling", USAEC Report AEC-tr-3770 (1952).
13. Zuber, N., "Hydynamic Aspects of Boiling Heat Transfer", USAEC Report AECU-4439 (1959).

REFERENCES (continued)

14. Corman, J.C., Walmet, G.E., "Vaporization from Capillary Wick Structures", ASME Paper 71-HT-35.
15. Basiulis, A., Filler, H., "Operating Characteristics and Long Life Capabilities of Organic Fluid Heat Pipes", AIAA Paper No. 71-408, 1971.
16. Ferrell, J.K., Johnston, H.R., "The Mechanism of Heat Transfer in the Evaporator Zone of a Heat Pipe", ASME Paper 70-HT/SPT-12, 1970.
17. Gregory, F.C., "An Investigation of Nucleate Boiling from Mesch Covered Surfaces", M.S. Thesis Naval Postgraduate School, June 1970 (AD 709 097).
18. Costello, C.P., Frea, W.J., "The Roles of Capillary Wicking and Surface Deposits in the Attainment of High Pool Boiling Burnout Heat Fluxes", AIChE Journal, Vol. 10, No. 3, p. 393, May 1964.
19. Tong, L.S., "Boiling Heat Transfer and Two-Phase Flow", John Wiley and Sons, 1965.
20. Cheng, S., "The Prediction of the Thermal Conductivity of Heterogeneous Mixtures", PhD Thesis, Auburn University, 1968.
21. Soliman, M.M., Graumann, D.W., Berenson, P.J., "Effective Thermal Conductivity of Dry and Liquid-Saturated Sintered Fiber Metal Wicks", ASME Paper 70-HT/SPT-40, 1970.
22. Zanker, A., "Thermal Conductivity Calculation of Porous Media by Means of Nomographs", Int'l. Jour. Heat and Mass Transfer, Vol. 13, p. 1359-63, 1970.
23. DeVries, D.A., "The Thermal Conductivity of Granular Materials", Annexe 1952-1 Bul. Inst. Intern. du Froid, p. 115-131, 1952.
24. Gorring, R.L., Churchall, S.W., "Thermal Conducitivity of Heterogeneous Materials", Chemical Engineering Progress, Vol. 57, July 1961, p. 53.
25. Krupiczka, R., "Analysis of Thermal Conductivity in Granular Materials", Int'l. Chemical Engineering, Vol. 7, No. 1, Jan. 1967.

REFERENCES (continued)

26. Busse, C.A., "Heat Pipe Thermionic Converter Research in Europe", 4th Intersociety Energy Conversion Engineering Conference, Sept. 22-26, 1969, p. 861.

27. Proceedings of Joint Atomic Energy Commission/Sandia Laboratories Heat Pipe Conference, SC-M-66-623, October 1966.

28. Harbaugh, W.E., "The Development of an Insulated Thermionic Converter-Heat Pipe Assembly", AF Technical Report AFAPL-TR-67-45, April 1967.

29. Cotter, T.P., "Heat Pipe Startup Dynamics", Thermionic Conversion Specialist Conference, Oct. 1967, p. 344.

30. Deverall, J.E., Kemme, J., Florschuetz, L.W., "Sonic Limitations and Startup Problems of Heat Pipes", Los Alamos Scientific Laboratory LA-4518 (1970).

31. Sockol, P.M., Forman, R., "Re-examination of Heat Pipe Startup", Thermionic Conversion Specialists Conference, Oct. 1970, p. 571.

32. Kemme, J.E., "Ultimate Heat Pipe Performance", IEEE Trans. on Electron Devices, Vol. ED-16, No. 8, Aug. 1969.

33. Levy, E.K. and Chou, S., "Vapor Compressibility Effects in Heat Pipes", AEC Report NYO-4095-1, May 1970.

34. Kemme, J.E., "Heat Pipe Capability Experiments", Los Alamos Scientific Laboratory Report LA-3585-MS.

35. Bähr, A., Burck, E., Hufschmidt, W., "Liquid-Vapor Interaction and Evaporation in Heat Pipes", Proc. Second Int'l. Conf. on Thermionic Electrical Power Generation, 1968.

36. Haskin, W.L., "Cryogenic Heat Pipe", AF Technical Report AFFDL-TR-66-228, June 1967.

37. Joy, P., "Optimum Cryogenic Heat Pipe Design", ASME Paper 70-HT/SPT-6.

38. Chi, S.W., Cygnarowicz, T.A., "Theoretical Analysis of Cryogenic Heat Pipes", ASME Paper 70-HT/SPT-6.

REFERENCES (continued)

39. Bienert, W., "Heat Pipes for Temperature Control", 4th Int'l. Intersociety Energy Conversion Engineering Conference, Sept. 1969, p. 1033.

40. Marcus, B.D., Fleischman, G.L., "Steady-state and Transient Performance of Hot Reservoir Gas-Controlled Heat Pipes", ASME Paper 70-HT/SPT-11, 1970.

41. Hinderman, J.D., Waters, E.D., Kaser, R.V., "Design and Performance of Non-condensable Gas Controlled Heat Pipes", AIAA Paper 71-420, April 1971.

42. Bienert, W., Brennan, P.J., Kirkpatrick, J.P., "Feedback Controlled Variable Conductance Heat Pipes", AIAA Paper 71-421, April 1971.

43. Feldman, K.T., Jr., Whitlow, G.L., "Experiments with a Two-Fluid Heat Pipe", 4th Intersociety Energy Conversion Engineering Conference, Sept. 1969, p. 1025.

44. Tien, C.L., "Two-Component Heat Pipes", AIAA 4th Thermophysics Conference, June 1969, p. 423.

45. Johnson, G.D., "Corrosion Studies of Liquid Metal Heat Pipe Systems at 1000°C to 1800°C", AIME Paper No. F69-2, 1969.

46. Busse, C.A., et al., "Heat Pipe Life Tests at 1600°C and 1000°C", IEEE Thermionic Conversion Specialist Conference, 1966, pp. 149-58.

47. Deverall, J.E., Kemme, J.E., "Satellite Heat Pipe", Los Alamos Scientific Laboratory, LA-3278-MS, April 1965.

48. Schwartz, J., "Performance Map of the Water Heat Pipe and the Phenomenon of Non-condensable Gas Generation", ASME Paper 69-HT-15, 1969.

49. Waters, E.D., King, P.P., "Compatibility Evaluation of an Ammonia-Aluminum-Stainless Steel Heat Pipe", ASME Paper 70-HT/SPT-15, 1970.

ATE

LMED

-8